

Design and manufacturing of a V-type Stirling engine with double heaters

İhsan Batmaz *, Süleyman Üstün

Gazi University, Faculty of Technical Education, 06500 Besevler, Ankara, Turkey

ARTICLE INFO

Article history:

Available online 22 April 2008

Keywords:

Stirling engines
Double displacer
Helium
Charge pressure

ABSTRACT

Under the consideration of the solar energy potential of Turkey, a V-type Stirling engine having two heaters was designed, optimized and then manufactured. The prototype engine was tested in laboratory condition using an electrical heating system. Tests were conducted within the temperature range of 650–1000 °C with 50 °C increments. The pressure ranged from the ambient value to 2 bar with 0.5 bar increments at each stage of temperature. The maximum power was obtained at 950 °C and 1.0 bar charge pressure as 118 W.

© 2008 Elsevier Ltd. All rights reserved.

1. Introduction

Stirling engines are externally heated, closed-cycle regenerative machines. Theoretical thermodynamic analysis of Stirling cycle engines indicates a thermal efficiency equal to that of the Carnot cycle. Although invented before internal combustion engines, Stirling engines are not yet commercially available today. The main reason preventing the Stirling engine from being commercial is its incompetitiveness with internal combustion engines in specific power and weight [1–3].

After inventing the Stirling engine in 1816, Robert Stirling carried out investigations up to 1850s and built double and triple cylinder engines, which were different in construction from the first engine. However, the performance of these engines was no better than that of the first engine. In 1860, Lehman built an horizontal single cylinder, displacer piston type Stirling engine without regenerator. This engine was found to be more popular than the regenerative engines built by Stirling [3,4]. In 1876, Rider used a novel construction on which expansion and compression spaces were set in different cylinders. On this engine, cylinders were present side by side and externally connected to each other by a regenerator. The superiority of this engine was the simple maintenance and silent operation as a result of having only five moving parts. Up to 1915, the only progress made on Stirling engines was the rearrangement of cylinders at right angles to each other by Robinson. The best Stirling engine manufactured by the end of 1923 had about 800 kg mass and 1.5 kW power output and 3% thermal efficiency [4,5].

In 1938, Professor Host noticed that there was a tremendous difference between the theoretical and the practical thermal efficiency of Stirling engines. Professor Host found this situation promising in consideration of the developments in material technology, and the research was started again in the Physical Research Laboratories in Eindhoven under the direction of H. Rinia. Several successful prototypes were built 50 times lighter in weight and 125 times smaller in volume than the engine designed in 1923. The results of their researches were published in 1947. These engines, called Philips Stirling Engines, worked under 20–50 bar charge pressure and had a maximum speed of 2000 rpm and an output of 2 HP [5,6].

In 1953, a new engine was invented by R.J. Meijer in Philips-Eindhoven, the Rhombic Drive Mechanism [7]. The primary advantage of this engine was that the piston applies no force to the cylinder wall and allows the use of a flexible seal between the piston and cylinder wall. Consequently, no charge pressure is needed in the crankcase. Meijer used hydrogen, helium and

* Corresponding author. Tel.: +90 312 202 86 45; fax: +90 312 212 00 59.
E-mail address: ibatmaz@gazi.edu.tr (İ. Batmaz).

air as working gases and obtained the best performance with hydrogen, the highest performance obtained up to that time [8].

After 1960, some engine companies (MAN-MWM, United Stirling of Sweden, Ford Motor Company of Detroit, etc.) started research programs to develop Stirling engines for automotive applications. Within these programs some multi-cylinder engines having different driving mechanisms were built and tested. The thermal efficiency of Stirling engines designed for automotive application was over 40%. However, the weights and volumes are not at desired levels [1].

In 1974, W. Beale invented the free-piston Stirling engine. In this engine, the phasing between working piston and displacer was not performed by a driving mechanism. Therefore, the engine had lower volume and weight, and thus, it had several advantages [9]. Particularly in solar energy research, it is appropriate to place the engine at the focus of a solar collector.

The practical cycle of Stirling engines differs from the theoretical cycle consisting of two constant temperatures and two constant volume processes. For the thermodynamic analysis of Stirling engines, more authentic approximations were devised. Schmidt [10] introduced one of the approximations with the assumption that the fluids in hot and cold spaces are at the same temperature as the hot and cold sources, respectively. In this analysis instead of theoretical processes, the real processes performed by pistons moving sinusoidally with 90-degrees phase angle were used. Another thermodynamic analysis was devised by Finkelstein [11], called nodal analysis, wherein the whole engine was divided into 13 sub-volumes. In this analysis, the temperatures at sub volumes are calculated with differential time intervals by the First Law of Thermodynamics. In the analysis, the instantaneous variations of the wall temperatures are also taken into account. Urieli et al. [12,13] improved the nodal analysis to cover the effects of leakage, heat losses and flow losses [14]. They conducted an investigation on nodal analysis, prepared computer programs and applied to the design of a rhombic driven engine for NASA. The details of nodal analysis can be found in their report.

In the present study, a V-type Stirling engine prototype was designed and manufactured for solar energy applications and for domestic applications. The reserve of consumable energy sources such as oil and natural gas has been decreasing, and cost input for production has been increasing rapidly in recent days. It is expected that these sources will be run out within 35–200 years [15]. In terms of solar energy, Turkey has a big potantice and solar energy is used for different purposes. In particular, during the summer time, solar energy per area is quite high [16].

2. Engine construction

The engine was designed as V-(α) type with two heaters (double displaces). In V-type engine, there is a 90° phase angle between cylinders. The 90° phase angle enables the processes of the cycle. Schematic picture of Stirling engine is shown in Fig. 2.1. The area is shown in Fig. 2.1 as number 2 is called hot area for heating of process material and also number 6 is called cold area for cooling of process material. In order to alienate process material easily and to raise difference between cold and hot sources, cold area was cooled by cold water. The system is completely closed. Helium was used as a process material. The process material which is heated in number 2 hot area generates output by expanding in number 10 cylinders.

2.1. Heater and cooler

Heater and cooler consist of four parts as displacer piston with center bearing, displacer piston, displacer cylinder, displacer connecting piston. Displacer piston provides to displace process materials (helium) by actuating between hot and cold areas, and meanwhile provide to transfer necessary heat energy to process materials with constant temperature in order to operate engine. Displacer piston operates in center bearing without lubrication. Center bearing was manufactured from hard cast iron with high graphite content in order to avoid inavodouble bearing friction. Displacer cylinder was manufactured from Cr alloyed steel with 3 mm thickness to rise its resistance against corrosion and heat. There is 1 mm gap between each displacer piston and cylinder. This gap also performs as regenerator. Heat is transferred from cylinder to process material by radiation. Whilst process material passes between piston and cylinder, its pressure and temperature is raised by absorbing heat and then output is generated. The temperature of displacer piston is less than that of cylinder surface and considering that while temperature of cylinder surface is at 800 K, temperature of displacer piston is 700 K. If the heat transferred by radiation is calculated between both surfaces, it is about 300 W. Some of this heat is transferred to process material and the rest is lost by different ways.

2.2. Cylinders and piston

Cylinder manufactured from high C cast iron is dry jacketed type. Cylinder jacket was driven into its hole with suitable tightness by pressing.

Power piston generated output by consuming energy load in itself with process material. Meanwhile power piston is moved linearly. This action is transmitted to crankshaft by means of piston rod. Power piston was manufactured from high C cast iron without ring in order to reduce friction. An inertial force was very high due to piston weight. Piston weight was reduced to 300 g. Then engine speed was increased. A piston extension was attached to the piston top in order to minimize dead volumes in engine.

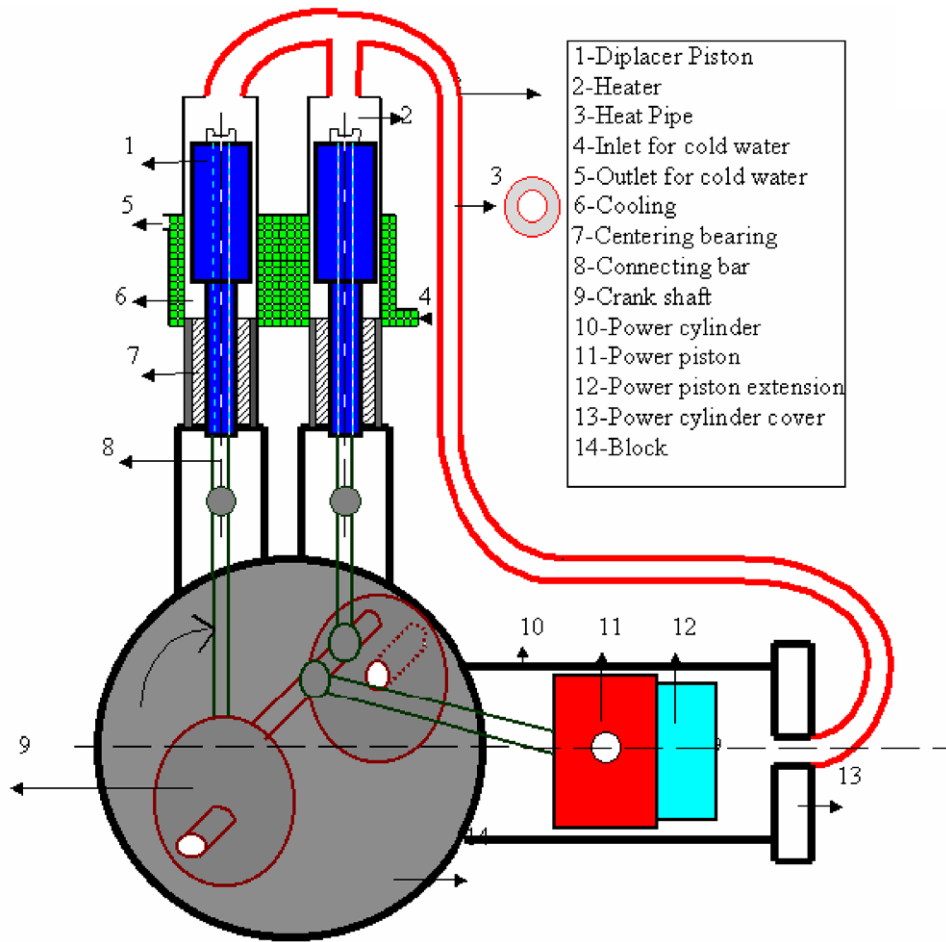


Fig. 2.1. Schematic drawing of manufactured Stirling engine with two displacer pistons.

3. Thermodynamic analyses

P - V and T - S diagrams of Stirling cycle are given in Fig. 3.1. Based on this cycle the operation of Stirling engines is explained through Fig. 3.2. The cycle consists of two isotherms (constant temperature) and two isochors (constant volume). The heat is given to process material which is placed in closed cylinder by means of heat exchanger heater. The exhausting of heat is performed by another heat exchanger cooler. Necessary heat for engine is provided from a special combustion chamber out of cylinder and a continuous combustion is maintained there.

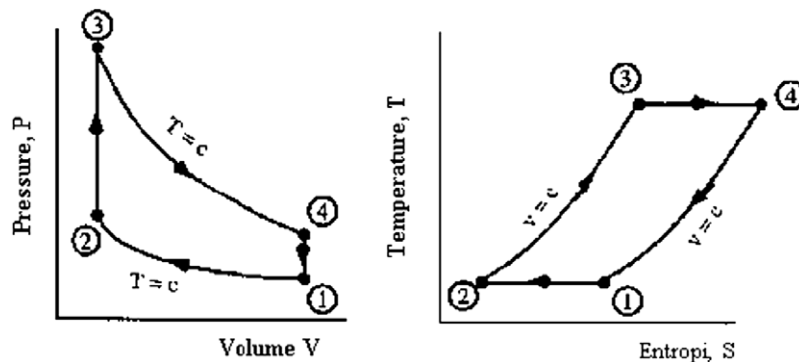


Fig. 3.1. P - V and T - S diagrams for Stirling engine.

3. There is a linear and stable temperature distribution in the stiff part of regenerator and it provides heat balance for the stiff part of regenerator.
4. Helium is considered ideal gas and used as a process material.
5. Total mass in the system does not change with time.
6. The walls of hot cylinder and heater are at hot source temperature and the walls of cold cylinder and cooler are at cold source temperature.
7. The volumes in heater, cooler and regenerator are constant; the volumes in hot and cold cylinders alter according to the crank angle, and they can be expressed by following formula:

$$V_C = V_D + \frac{1}{2} V_S [1 + \cos(\theta - \phi)] \quad (1)$$

$$V_H = V_D + \frac{1}{2} V_S [1 + \cos(\theta)] \quad (2)$$

There are three unknown terms such as pressure, temperature and density and the formula which can be used to determine the above terms can be given as follows based on formula for ideal gases, mass balance and energy balance, respectively:

$$p = \rho RT \quad (3)$$

$$m_1 + m_2 + m_3 + \dots + m_n = m_t \quad (4)$$

$$dQ - dW + dH_{\text{inlet}} - dH_{\text{outlet}} = dU \quad (5)$$

Taking into consideration that pressure is the same at every condition and (3) and (4) can be evaluated as follows:

$$p = \frac{m_t R}{\frac{V_1}{T_1} + \frac{V_2}{T_2} + \frac{V_3}{T_3} + \dots + \frac{V_n}{T_n}} \quad (6)$$

The formula given below can be arranged according to energy balance:

$$h_i A_i (T_{wi} - T_i) dt - p dV_i + (dH_{\text{inlet}} - dH_{\text{outlet}}) = m_i C_v dT_i + C_v T_i dm_i \quad (7)$$

where $(dH_{\text{inlet}} - dH_{\text{outlet}})$ term in the formula presents input and output energy from component bar dens and is calculated as follows:

$$(dH_{\text{inlet}} - dH_{\text{outlet}})_i = C_p [(m_1^t + m_2^t + \dots + m_{i-1}^t) - (m_1^{t+\Delta t} + m_2^{t+\Delta t} + \dots + m_{i-1}^{t+\Delta t})] \left[\frac{T_{i-1}}{T_i} \right] + C_p [(m_{i+1}^t + m_{i+2}^t + \dots + m_n^t) - (m_{i+1}^{t+\Delta t} + m_{i+2}^{t+\Delta t} + \dots + m_n^{t+\Delta t})] \left[\frac{T_i}{T_{i+1}} \right]$$

In the above formula, the first term shows the energy which enters or leave from the component from the left side; the second term shows the energy which enter to or leave from the component from the right side. The temperatures are written in matrix column; those are located upper are used for flow which moves from left to right and those located lower are used for flow which moves from right to left. The formula used numerical procedures as follows [17]:

$$\Delta T_i = [h_i A_i (T_{wi} - T_i) \Delta t - (m_i^{t+\Delta t} - m_i^t) C_v T_{i+}] \quad (8)$$

$$(dH_{\text{inlet}} - dH_{\text{outlet}})_i - P \Delta V_i / C_v m_i$$

P–V diagram which is designed and obtained by software by using this analysis method is shown in Fig. 3.4.

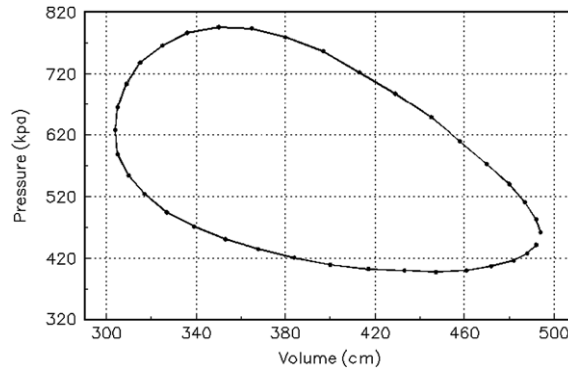
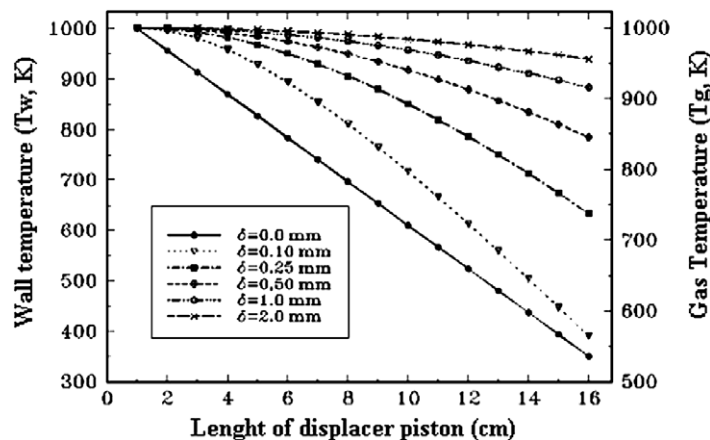


Fig. 3.4. P–V diagram obtained by software.

Table 3.1

Calculated and measured engines values

Properties	Obtained by software	
Type of engine	V (α)	
Hot cylinder volume (V_E)	311 cm ³	
Cold cylinder volume (V_H)	311 cm ³	
Total dead volume (V_D)	290.57 cm ³	
Diameter of cylinder (D)	61 mm	
Piston stroke (L)	65 mm	
Fazes angle (ϕ)	90°	
Cold source temperature (T_C)	350 K	318 K
Hot source temperature (T_H)	1273 K	953 K
Maximum engine speed (n_{max})	1200 l/min	784 l/min
Carnot cycle (η_C)	% 72 ^a	% 0.66
Net work (W_{net})	245 J	
Engine efficiency (η)	11%	
Charge pressure (P_{charge}), (manometric)	2 bar	2 bar
Max cycle pressure (P_{max})	3.048 bar ^a	
Mean cycle pressure (P_{ort})	2.810 bar ^a	
Process material	Helium	
Output (N)	500 W	118 W
Compression ratio (ϵ)	1.65 ^a	
Minimum volume (V_{min})	290.57 cm ³	
Maximum volume (V_{max})	480.53 cm ³	

^a Value based on theoretical calculation.**Fig. 3.5.** Variation between gas and wall temperatures depending on the length of displacer piston.

The values belonging to Stirling engine by using software are shown in Table 3.1.

The manufactured engine was designed for 500 W output. However 118 W of output was obtained during the tests. One of the reasons for low output is access section distance (δ). Since the access section gets more narrow and since gas temperature gets closer to wall temperature, engine efficiency will rise, if the distance is far away as much as possible between hot and cold source according to Carnot cycle. On the contrary, the more the access section distance rises, the less the differences between cold and hot sources, hence the efficiency gets lower.

Different access section with various gas temperatures and wall temperatures are shown in Fig. 3.5 based on the length of the displacer piston. As it is seen in this figure, wall temperature and gas temperature vary linearly depending on the length of the displacer piston. However, since the distance between displacer piston and cylinder (access sections) reduces, the better heat transmission is obtained; gas temperature gets closer to wall temperature. On the other hand, even it is clear that when the access section decreases more, consequently heat transmission gets well, since access of process material becomes difficult; losing output (pumping work) gets increased. When taking manufacturing into account, it is difficult to centralize displacer piston, if the access section is very small. If the access section distance is very high, heat transmission gets worse; therefore, gas temperature recedes from wall temperature. The access section is given 1 mm width in manufacturing.

4. Engine performance

Test engine was operated at constant oven temperatures without charge, and the change of speed, torque, output vibration was measured in the same conditions, but at the different charge pressure speed, torque, output and vibration were

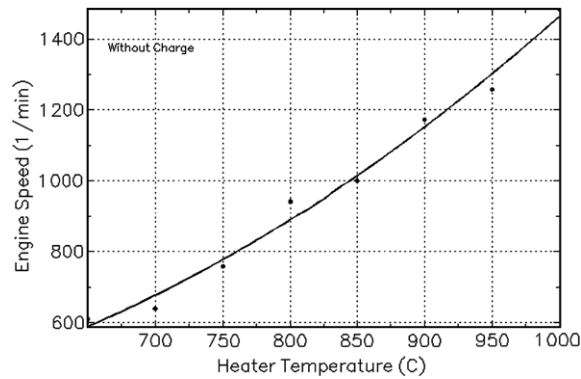


Fig. 4.1. Variation of engine speed depending on heater temperature (without charge).

examined (gauge pressures of 0.5 bar, 1.0 bar, 1.5 bar, 2.0 bar). Fig. 4.1 shows the effect of heater temperature on engine speed.

Variation of engine speed depending on heater temperature is shown in Fig. 4.1. As it is seen in this figure, while heater temperature increases, engine speed also increases linearly. Engine speed varies based on heater temperature. In Fig. 4.2 variations of engine speed depending on heater temperature are shown at different charge pressures. As it is seen in this figure, the engine speed of 784 l/min was reached at 10 bar pressure, 1000 °C, and speeds vary depending on heater temperature. Engine speed of 645 l/min was reached under the atmospheric condition at 1000 °C (Fig. 4.1). The engine speed was high at charge conditions compared to atmospheric conditions, owing to the increase of mass of process materials.

4.1. Impact of heater temperature on torque and output

The variation of engine torque and output depending on heater temperature is shown in Fig. 4.3, output of 34 W was obtained at 950 °C. A graph was created by taking maximum torque and output values for each heater temperature; increasing of torque and output depending on heater temperatures causes different temperatures between heater and cooler. Owing to heater temperature increase up to material endurance limit raises cycle output and engine efficiency, torque and output also rise depending on it.

The variation of engine torque and output at different charge pressures and heater temperatures is shown Fig. 4.4. Maximum power of 118 W was obtained at 950 °C and at 1.0 bar pressure. When looked at torque curve, the same situation as power curve can be seen. The reason for decrease of torque and output after 1.0 bar is due to the inadequacy of temperature heat transfer surface area.

5. Discussion and outlook

Engine efficiency was 11%. It is necessary to eliminate dead volumes and leakage in the system which increases engine efficiency.

Engine speed increases depending on heater temperature. Increasing of heater temperature up to material endurance limit was very effective in improving engine torque and output.

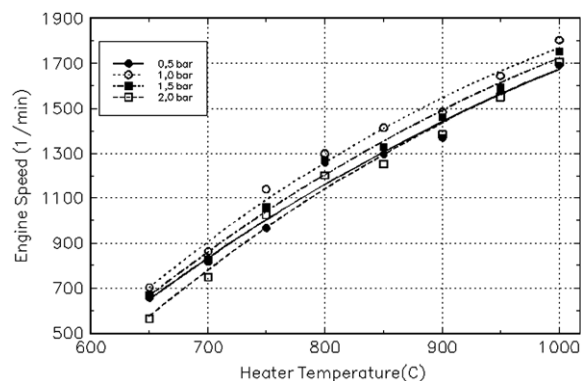


Fig. 4.2. Variation of engine speed depending on heater temperature (with charge).

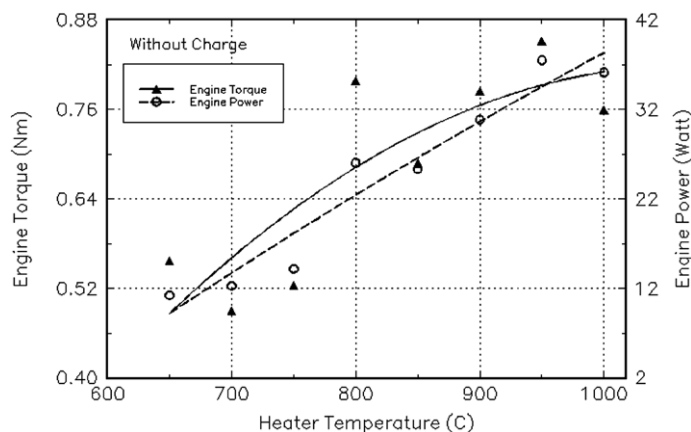


Fig. 4.3. Variation of torque and output depending on heater temperature (without charge).

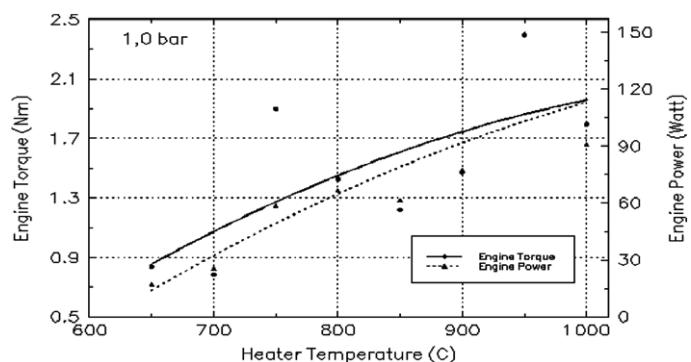


Fig. 4.4. Variation of torque and output depending on heater temperature (at 1 bar pressure).

Charge pressure to system was performed from carter. In order to decrease the leakage, it will be high efficiency, if the charge performs only from the heater and chiller sections.

In this study, it was designed to obtain 500 W of output, but output of 118 W was obtained at 594 l/min during the tests. In order to take more output, it is necessary to minimize dead volumes, to well balance between crank and rod mechanism, and provide very good leakage to charge the system.

Another use which needs to solve is to make Stirling engine compete with internal combustion engine rather than leakage, dead volumes and improvement of heat transmission, storage of solar energy. Due to solar energy being familiar to environment and very cheaper, same collectors which are used to collect solar energy can be used so that Stirling engines can be used for irrigation purpose.

Manufactured V-type engine with double displacer pistons is an example which is known in Turkey. Prototype balancing, leakage, heat transmission and dead volumes still exist. If these problems are solved, they can be used as stationary plant engine in the market.

References

- [1] Walker G. Stirling engines. Calredon Press Oxford; 1981.
- [2] Pavletic R, Prebil I, Vuksanovic B. Better cycle engines. Yugoslavia: University of Ljubljana; 1991.
- [3] Rix DH. Some aspects of the outline design specification of a 0.5 kW Stirling engine for domestic scale co-generation. J Power Energy, IMechE 1996;210:25–33.
- [4] Finkelstein T. Air Engines II. The Engineer 1959(April 3):522–7.
- [5] Finkelstein T. Air Engines III. The Engineer 1959(April 10):568–71.
- [6] Urieli I, Rallis CJ. Stirling cycle engine development—a review. Energy Utilization Unit Paper No. 7–5. University of Cape Town; 1975.
- [7] Finkelstein T. Air Engines IV. The Engineer 1959(May 8):720–3.
- [8] Meijer RJ. The Philips Stirling thermal engine. Thesis, Technische Hogeschool Delft, November 1960.
- [9] Michels APJ. The Philips Stirling engine: a study its efficiency as a function of operating temperatures and working fluids, 769258. In: 11th IECEC, Sahara Tahoe Hotel, Statline, Nevada, September 12–17 1976.
- [10] Beale W, Holmes W, Lewis S, Cheng E. Free-piston Stirling engine—a progress report, SAE, Paper no. 730647, 1973.
- [11] Schmidt C. The theory of Lehmanns calorimetric machine. Zeitschrift des Vereines Deutcher Ingenieure 1871:15. Part 1.
- [12] Finkelstein T. Thermodynamic analysis of Stirling engines. J Spacecraft Rockets 1967;4(9):1184–9.

- [13] Urieli U, Berchowitz DM, Stirling cycle machine analysis. Bristol: Adam Hilger Ltd.; 1984.
- [14] Urielli I, Kushnir M, The ideal adiabatic cycle—a rational basis for Stirling engine analysis. In: 17th IECEC, Westin Bonaventure, Los Angeles, CA, 8–12 August 1982.
- [15] Tew R, Jefferies K, Miao D, A Stirling engine computer model for performance calculations, Report No. DOE/NASA/1011-78/24, July 1978.
- [16] Uyarel AY, Öz ES, Güneş Enerjisi ve Uygulamaları, Emel Matbacılık Sanayi, Ankara; 1987.
- [17] Karabulut H, Stirling Motorlarının Termodinamik Similasyonu, Türk Isı Bilimi ve Tekniği Dergisi, Mart, Ankara; 1997.